DEPARTMENT OF THE ARMY
CORPS OF ENGINEERS

Technical Report 1695-TR

AIR CONDITIONERS, 9,000-BTU/HR, MULTI-PACKAGE, 115-VOLT 60-CYCLE AND 208/416-VOLT 400-CYCLE 4-WIRE

Task 8F71-11-001-03

11 September 1961

U S Army Engineer Research And Development Laboratories

FORT BELVOIR, VIRGINIA

U. S. ARMY ENGINEER RESEARCH AND DEVELOPMENT LABORATORIES
CORPS OF ENGINEERS

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The Director
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Prepared by

Robert L. Little
Special Equipment Branch
Civil Department
U. S. Army Engineer Research and Development Laboratories
Corps of Engineers
Fort Belvoir, Virginia

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PREFACE

Authority for the development described in this report is contained in Task 8F71-11-001-03 (formerly Project 8-71-11-400 and 8-88-06-103). A copy of the project card is included in Appendix A.

Development and tests were conducted under the supervision of three consecutive project engineers: George W. Andrews, Hugo R. Lopez, and Robert L. Little.

CONTENTS

SUMMARY

This report covers the testing and development of two 9,000-Btu/hr, multi-package (sectional) air conditioners available in two power versions: 60-cycle and 400-cycle. The units were developed under Contract DA-44-009 Eng-3320 with Ellis and Watts Products, Inc., Cincinnati, Ohio. Capacity, air flow, and environmental tests are described.

The report concludes:

a. The 60-cycle and 400-cycle, 9,000-Btu/hr, multi-package air conditioning units do not meet all requirements set forth in the purchase description.

b. The 60-cycle and 400-cycle units are suitable for applications where the multi-package feature is required.

c. There is no necessity for inclusion of the 416-volt operating requirement in design of the units.

d. The hose storage compartment on the unit is an unnecessary waste of space.

e. The multi-package design could be modified:

(1) To result in dimensions which more nearly correspond to those of the single-package unit.

(2) To eliminate the hose storage compartment.

(3) To eliminate the 416-volt components, controls, and wiring.

(4) To incorporate the "Tablock" fan in the evaporator section for increased air flow and sensible heat factor.

(5) To incorporate copper connecting lines in lieu of flexible hoses wherever installations will permit.

AIR CONDITIONERS, 9,000-BTU/HR, MULTI-PACKAGE, 115-VOLT 60-CYCLE AND 208/416-VOLT 400-CYCLE 4-WIRE

I. INTRODUCTION

1. Subject. This report covers the development and test of two 9,000 British thermal units per hour (Btu/hr), multi-package (sectional) air conditioners. These multi-package units (which permit mounting the evaporator section inside the conditioned space and the condenser section outside the conditioned space) were developed for use in vans and shelters where it is not feasible to cut large areas out of the wall.

2. Background and Previous Investigations. Although a single-package, 9,000-Btu/hr, 12-volt, single-phase, 60-cycle air conditioner already existed as a standard item, both the Signal Corps and the Ordnance Corps were interested in obtaining a unit which would not require that a 16- by 26-inch mounting hole be cut in their vans and shelters since this weakens the wall structure. When a multi-package unit is used, the evaporator section can be mounted in the conditioned space and the condenser section can be mounted outside. With this arrangement, a hole only 2 or 3 inches in diameter is needed to pass the refrigerant and electrical lines through the wall.

The 9,000-Btu/hr, multi-package air conditioners were developed by Ellis and Watts Products, Inc., Cincinatti, Ohio, under Contract DA-44-009 Eng-3320 awarded 6 June 1957. Four prototype units were to be delivered by 6 February 1958. Delivery was delayed numerous times by difficulties in obtaining components, strikes by component suppliers, and failures of the unit to meet requirements of the purchase description. The first prototype was delivered in August 1958, and the last was accepted in early 1960.

II. INVESTIGATION

3. Description. The 9,000-Btu/hr multi-package air conditioner (Figs. 1 and 2) is 16 inches high, 26 inches wide, and 36 inches deep. The evaporator section (Fig. 3) is 16 by 26 by 16 inches, and the condenser section (Fig. 4) is 16 by 26 by 20 inches. The frame of each section is a 1- by 1- by 1/8-inch aluminum angle construction on the top and sides of each surface. The bottom of each section is a 1/8-inch aluminum plate with a 3/4-inch flange turned down to form an inverted pan. All pieces of the frame are welded together. Each section is equipped with removable angles attached to the back vertical edges so that the two sections can be joined back to back. Each unit is also supplied with flexible

refrigerant and electrical lines long enough for the two sections to be separated 4 feet. Each refrigerant line is equipped with quick-disconnect, self-sealing couplings. The control panel on the evaporator section is adapted for remote location with a 4-foot extension cable. The weight of one complete unit (both sections) is 215 pounds. The unit was developed in two different power versions: 115-volt, single-phase, 60-cycle power; and 208/416-volt, 3-phase, 4-wire, 400-cycle power.

The refrigerant cycle of the unit is the standard vapor cycle using refrigerant R-12.

The air-handling system consists of two fans, a filter, a set of dampers, and an adjustable discharge grille. The only air-handling component in the condenser section is the condenser fan. It is a propeller-type, direct-drive fan. The motor has sealed, long-life bearings and requires no lubrication. Fan operation is controlled by a temperature-actuated automatic switch. Located in the evaporator section are the second fan, filter, dampers, and louvers. The evaporator fan is a centrifugal, direct-drive type. This fan pulls room air through the return- and/or fresh-air damper and through the air filter and evaporator coil and forces it out the discharge louvers. The rated fan capacity is about 300 cfm of standard air. The return- and fresh-air dampers are connected by a mechanical linkage and are adjusted manually from 0 to 100 percent fresh air. The return-air damper is located on the front panel, and the fresh-air damper is on the rear. The air filter is a permanent type which may be cleaned by steam or warm water or by soaking in a solvent. A filter fluid should be applied after each cleaning.

The electrical controls of the 60-cycle unit utilize the following components for operation: wiring, switch, contactor, capacitors and starting relay, thermostat, and overloads. Each wire in the unit is number coded and may be traced by reference to the wiring diagram. The main switch, located on the control panel in front of the unit, is constructed to act as a manual "step starting" device. To keep high starting current to a minimum and to prevent circuit overloads, the operator should pause between each position to allow each major component to start separately. The switch has three positions: off, ventilating, and cooling. The off position electrically disconnects all power from the control circuit. The ventilating position provides constant operation of the circulating fan only. The cooling position provides automatic operation of the compressor and condenser fan and continuous operation of the circulating fan. A temperature-sensing control, located on the compressor discharge line, causes the condenser fan to stop if the refrigerant temperature becomes too low. The only contactor in the electrical circuit is located inside the rear evaporator panel. The contactor, used to open and close the compressor circuit, is energized by the

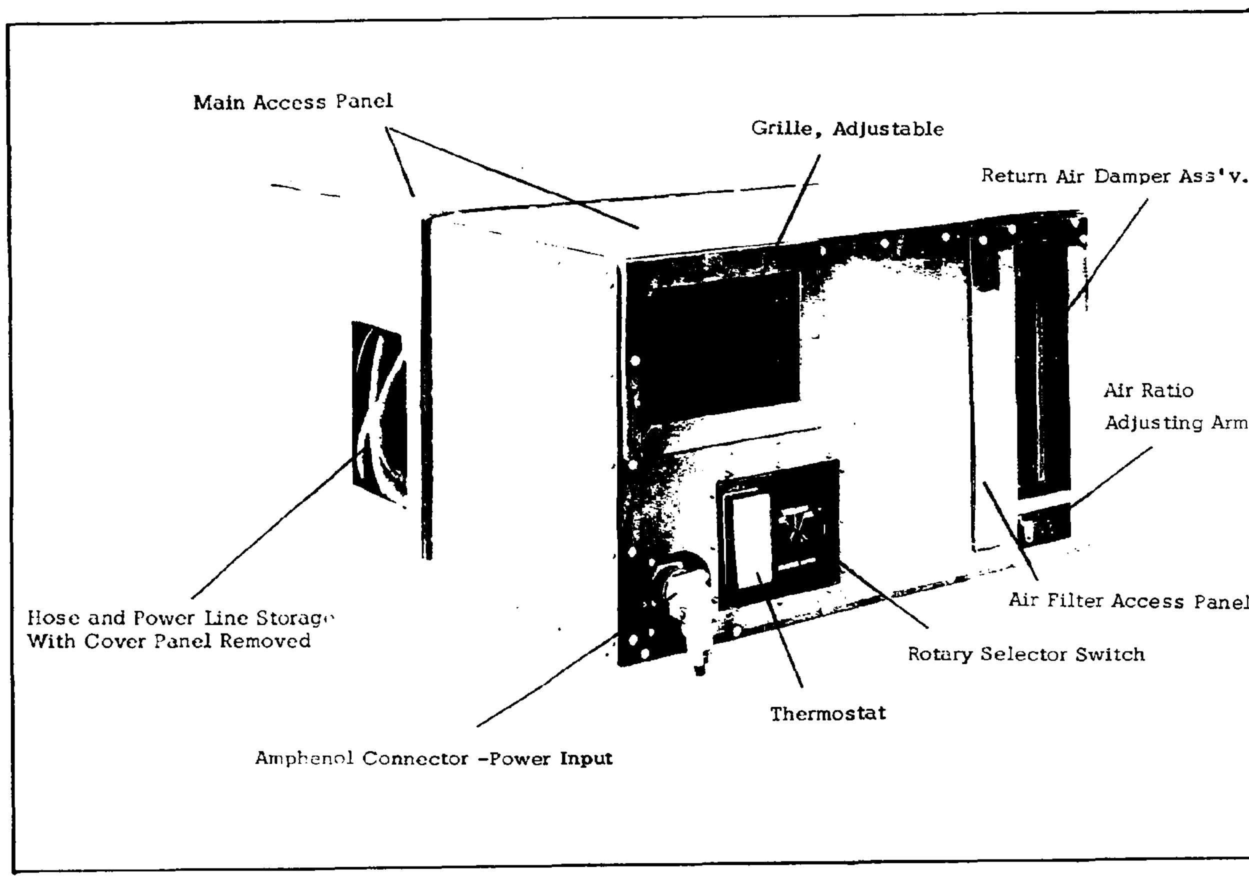

Fig. 1. Front view of Corps of Engineers model CE-9-B-60 air conditioning unit.

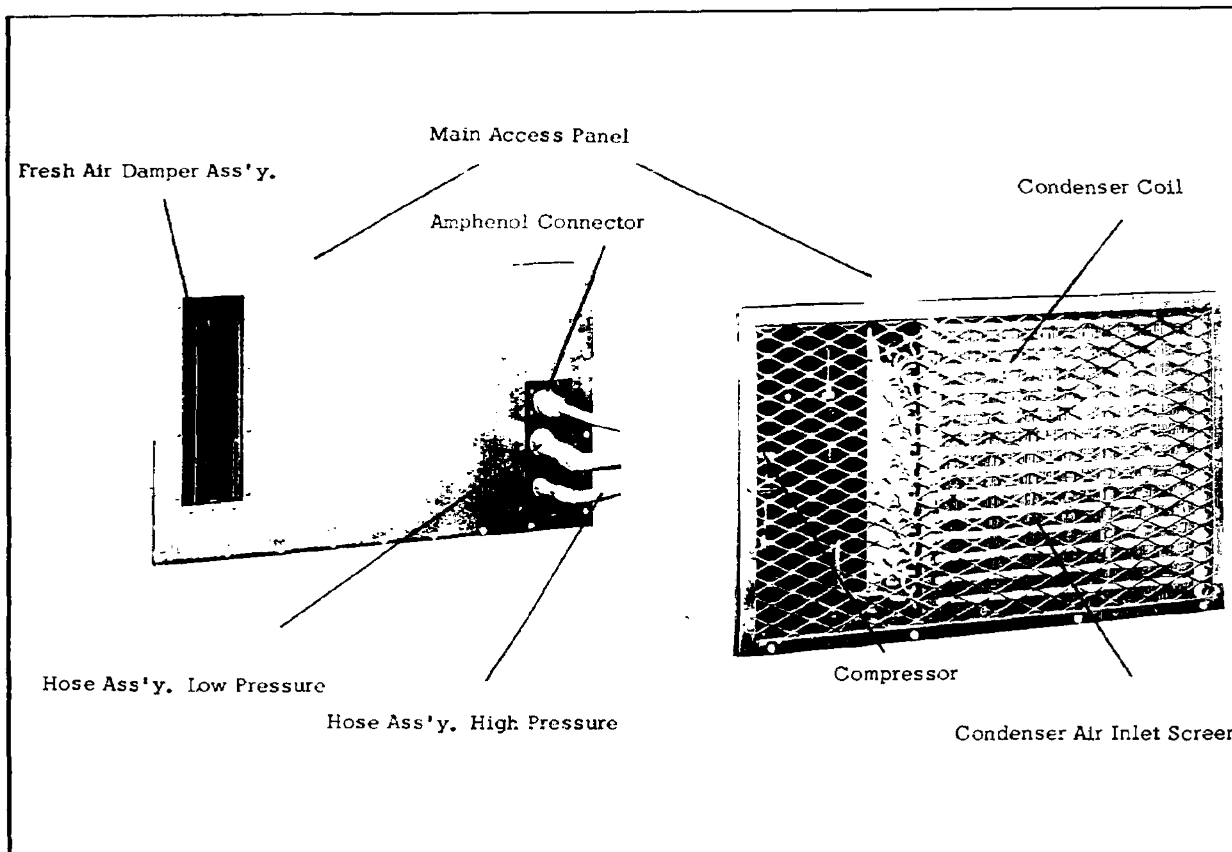

Fig. 2. Rear view of Corps of Engineers model CE-9-B-60 air conditioning unit.

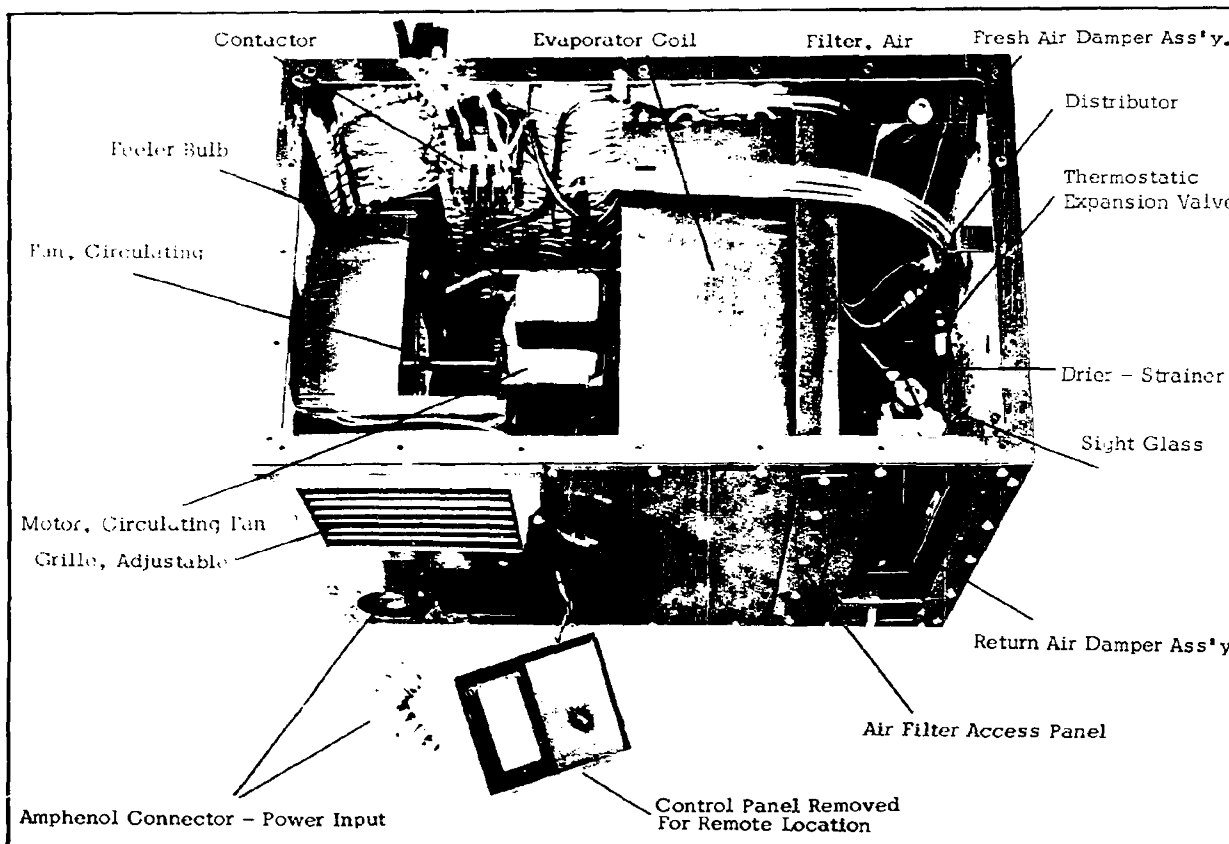

Fig. 3. Evaporator section of Corps of Engineers unit.

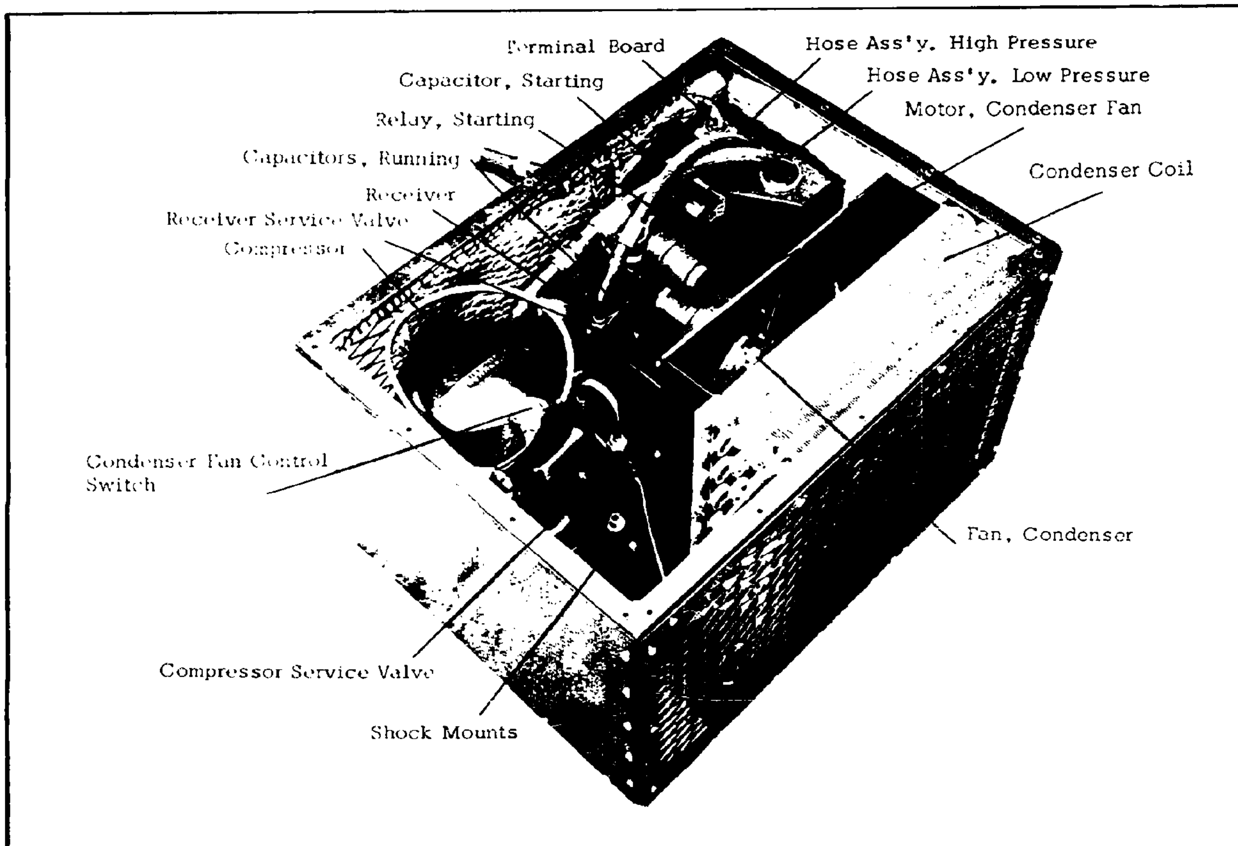

Fig. 4. Condenser section of Corps of Engineers unit.

unit's automatic and/or manual controls. Starting and running capacitors and a starting relay are needed to supply the power required for favorable compressor operation. The thermostat is located on the control panel beside the selector switch. Its function is to control compressor operation, depending on temperature of the conditioned space. Each motor in the unit is supplied with a thermal overload. The overloads are the automatic-reset type and are located inside the respective motors.

The wiring and controls of the 400-cycle unit do not differ greatly from those of the 60-cycle unit. The control circuit utilizes d-c current obtained from a transformer and rectifier. The 400-cycle unit also uses three contactors instead of the one required for the 60-cycle unit. The 400-cycle motors used in this design require additional capacitors to raise the power factor.

4. Performance Requirements. The 9,000-Btu/hr multi-package air conditioner was designed to meet the following performance requirements:

a. 9,000 Btu/hr net cooling capacity at an ambient temperature of 125° F dry bulb and 85° F wet bulb and air entering the evaporator at 90° F dry bulb and 75° F wet bulb.

b. 9,000 Btu/hr net cooling capacity at an ambient of 95° F dry bulb and 75° F wet bulb and air entering the evaporator at 80° F dry bulb and 67° F wet bulb.

c. Sensible heat ratio of from 68 to 80 percent.

d. Air flow of 300 to 350 cfm to the space being cooled and a minimum of 85 percent free air against 0.15 inch of water external static pressure.

e. An operating economy of 5.0 Btu/watt for the 60-cycle unit and 4.5 Btu/watt for the 400-cycle unit.

f. Condensing pressure not exceeding 245 psig.

g. Suction pressure not to drop below 45 psig.

h. Icing of evaporator coil not to occur when unit is operating in ambients as low as 65° F.

i. Shocks normally encountered in military mobile vans or trucks not to be damaging to the unit. (Shocks of 15 G in horizontal and vertical planes with a time duration of 11 milliseconds and vibrations of 3 G with a varying frequency of 10 to 400 cycles for 90 minutes and at resonant frequency for 90 minutes will be incorporated in the unit design.)

j. Range of thermostat to be 70° F to 90° F.

k. Maximum current draw for the 60-cycle model to be 15 amps.

l. Minimum power factor for the 400-cycle model to be 0.75.

m. Unit capable of withstanding storage at any ambient temperature from minus 80° F (for periods of at least 1 day duration) to plus 160° F (for periods as long as 4 hours per day).

n. Hermetic compressor having minimum life expectancy of 5 years.

o. Motors equipped with thermal overload protection.

5. Scope of Test Program. The tests described below are ones which are encountered in a complete testing program for air conditioners.

a. Capacity. Capacity tests are run at high and low ambient temperatures with normal atmospheric pressure and at a simulated high ambient temperature with pressure at 5,000 feet of altitude. The purpose of the test is to determine the net cooling capacity in British thermal units per hour.

b. Air Flow. The air flow test is performed on the evaporator fan at two conditions: First, with the evaporator coil dry (compressor not operating) and, second, with evaporator coil wet (compressor operating). The purpose of this test is to determine the quantity of air flow delivered by the unit.

c. Tilted Operation. The unit is tilted in four different positions, 90 degrees apart, at a slope of 5 degrees from horizontal. The unit is then started and operated long enough to determine whether or not proper condensate flow is being obtained.

d. Environmental Exposure. Included in the environmental exposure tests are high and low temperature, humidity, altitude, salt spray, vibration, rain, and shock tests. The purpose of each test is to determine the effect of different environmental conditions on the appearance and operation of the unit.

e. Radio-Interference Suppression. One 60-cycle model and one 400-cycle model are tested by the Government to determine if the interference limits to communication equipment are exceeded.

f. Controls. Each control of the unit is operated a sufficient number of times to insure that it functions as intended.

6. Conduct of Tests. Tests were started on the 60-cycle model in August 1958 at USAERDL. Tests were started on the 400-cycle model in August 1959. Capacity and air flow tests have been completed on both models, and vibration tests have been performed on the 60-cycle model. The remaining environmental tests have not been performed or witnessed by USAERDL personnel. Capacity and air flow tests were performed on one unit each of the 60- and 400-cycle models by Electrical Testing Laboratories (ETL) of New York. Results of these capacity tests are included in this report. Ellis and Watts conducted complete tests on a 400-cycle unit which was loaned to them by USAERDL as a preproduction test model under a Chicago Procurement Office contract. A copy of the Ellis and Watts report is on file at USAERDL. Shock and vibration tests were contracted to Inland Testing Laboratories, Dayton, Ohio, by Ellis and Watts. Ellis and Watts submitted a certified report (Appendix D) covering these tests to USAERDL.

a. Capacity. Test methods used for capacity tests were based on ASHRAE (ASRE) Standard 16-56, Methods of Rating and Testing Air Conditioners, published by the American Society of Heating, Refrigerating and Air Conditioning Engineers (ASRE). At USAERDL, the 60-cycle unit capacity tests were performed in a calibrated room-type calorimeter. Tests of the 400-cycle unit were first conducted using a simulated balanced ambient room-type calorimeter for the roomside calorimeter and a calibrated room-type outside calorimeter. After results of the first 400-cycle tests at USAERDL were found to be unfavorable, the tests were rerun using an attached psychrometric calorimeter method on the evaporator side. The ETL capacity tests utilized the balanced ambient room-type calorimeter method for both the 60- and 400-cycle units. The test methods described below are those of the ETL capacity tests.

Under the ASRE testing procedure, the sensible heat and latent heat removed from the air in the indoor test room by the unit under test were replaced by the indoor calorimeter air reconditioner which, at ETL, used electric strip heaters and an evaporator tray filled with water also electrically heated. When the wet-bulb and the dry-bulb thermometers located in the stream of sampled air taken 6 inches in front of the reconditioner outlet showed stable thermal equilibrium at specified conditions, the total heat input into the indoor reconditioner during the test period of one hour, modified by any heat leakage through the enveloping surfaces of the test room, was the cooling capacity of the unit under test.

As a check, the cooling capacity was again determined from the outdoor test room, into which was injected all the heat removed from the indoor test room by the unit under test. The latent heat and sensible heat introduced into the outdoor room were removed by the outdoor calorimeter air reconditioner which used heat transfer coils through which chilled water circulated. Electric strip heaters were provided in the reconditioner to reheat the dehumidified air as needed. Stable thermal equilibrium in this project (because of the physical arrangement of the units tested) was indicated by the thermocouples located in the stream of air taken 1 inch in front of the condenser inlet grille of the test unit. The total heat removed by the reconditioner during the test period of one hour, reduced by the thermal equivalent of the electrical input to the unit under test and to the reconditioner and also modified by any heat leakage through the enveloping surfaces of the test room, was the check cooling capacity of the unit under test. The ASRE Standard specified that the capacities determined from the two calorimeters shall agree within 6.0 percent.

The electrical inputs to the unit under test and to the two test rooms were measured by recently calibrated instruments (voltmeters, ammeters, and wattmeters) and energy input by integrating watthour meters. The quantity of water evaporated in the indoor room was measured by a modified hook gauge used in conjunction with a vernier scale which indicated the water level in the evaporator tray. Condensed water was collected from the evaporator section of the test unit and measured (the units do not discharge condensate to the condenser). The quantity of chilled water circulated in the outdoor reconditioner was measured with precision water meters. Dry-bulb and wet-bulb air temperatures and water temperatures were measured with precision glass thermometers graduated to 0.1° F.

The test rooms were each 12 feet long (parallel to the wall separating the two calorimeters), 8 feet wide, and 8 feet high. They were each surrounded by an interspace. The separating wall, which separated the interspaces as well as the test rooms, was insulated with 8-inch-thick blocks of cellular material known as "Foamglass." The walls, ceilings, and floors were completely covered with aluminum-sheathed panels which were impervious to vapor. The floors, in addition to the aluminum panels, were covered with 1/8-inch-thick linoleum. The window in the separating wall where the air conditioner under test was mounted was of the standard double-hung type suitably equipped with insulating stripping to reduce heat and moisture transfer at points of contact with the air-conditioner housing. An opening with sheet-plastic damper was provided on top of the window to equalize the pressure between the two calorimeters. The air static-pressure difference between the two calorimeters was maintained below 0.005 in. H_2O and was measured by an inclined manometer.

The interspaces were maintained at the temperatures of their respective test rooms (where feasible, otherwise measured) to minimize heat transfer. The temperatures of the air at each of the five surfaces of each test room exposed to the interspaces were measured by resistance thermometers and were recorded by a calibrated automatic multipoint recorder at $7\frac{1}{2}$-minute intervals.

For the test, the air conditioner was mounted with the evaporator section of the unit extending into the indoor test room. Cloth tape, felt, and foam-rubber strips were used around the side panels to guard against leakage. The air-discharge deflection louvers were set to blow the air away from the air-intake of the unit. The test-unit controls were set for maximum cooling with ventilation door closed.

Refrigerant temperatures were measured by thermocouples in conjunction with direct-reading, temperature-measuring instruments. Pressure of the refrigerant circuit was arrived at by conversion of temperature data to pressure by standard published tables. Where gauges are provided, the pressures are read directly. Condition of the refrigerant was evaluated by visual observation of the indicators of moisture and fullness of charge provided in the sight glass. Electrical characteristics were continuously recorded and evaluated during each operational test by use of accurately and recently calibrated instruments.

b. Air Flow. Air flow measurements of the evaporator section were performed with a small code tester. The code tester consisted of a mixing section, nozzle section, adjustable damper section, and a fan discharge section. All sections were fitted into a square duct. The mixing section consisted of a series of vanes arranged to divide the air flow into a number of small streams and then divert the streams across each other. The vanes extended only halfway across the duct and were positioned at approximately 45 degrees. The code tester contained two mixer assemblies, one to mix vertically and the other to mix horizontally. The condition of air passing through the nozzle was determined from readings of wet bulb and dry bulb thermometers located between the mixer and the nozzle section. The nozzle section was adapted so that two different nozzles could be fitted into the code tester. Both available nozzles were made to Bureau of Standards elliptical nozzle specifications. Located before and after the nozzle are static pressure taps for determination of pressure drop across the nozzle. Provisions for a Pitot tube were also available for determination of the velocity pressure through the nozzle. The adjustable damper section consisted of an adjustable iris which could be closed to increase the static pressure at the transition to the test unit. The fan discharge section consisted of a fan and discharge opening to the atmosphere. The fan speed was varied by changing pulleys.

The first air flow test on the 60-cycle model was performed using a new larger Pitot tube than had been used previously. The smaller of the two nozzles available was also used during the first test. When the results of the first run were unfavorable, the old Pitot tube was installed in place of the new one. Desired results were still not obtained; therefore, the larger nozzle was used in the next run. Results of tests run with the old Pitot tube and large nozzle were acceptable. Air flow tests on the 400-cycle model were performed using the small nozzle and the pressure taps to measure the pressure drop across the nozzle rather than the Pitot tube to measure velocity pressure. A new type of fan ("TABLOCK" by Torrington) was installed in the 60-cycle unit in an attempt to obtain greater air flow for the same size fan. The only test run on the new fan wheel was with a dry evaporator coil. The test method remained the same. The sequence of the air flow tests on each model of the 9,000-Btu/hr multipackage unit was the same. The units were first tested with a dry evaporator coil and then with a wet coil. The test units were allowed to run for a period of time to allow supply air conditions to stabilize. Data collection was started with a static pressure of 0 inch of water at the transition between the code tester and the unit. The pressure at this point was increased 0.05 inch of water for each subsequent reading until the adjustable iris was fully closed or until the fan on the test unit stalled out. The following data was recorded at each static pressure:

(1) Static pressure at transition duct - inches of water.

(2) Pressure drop across nozzle or velocity pressure at nozzle - inches of water.

(3) Temperature of air at nozzle - dry bulb - ^{o}F.

(4) Temperature of air at nozzle - wet bulb - ^{o}F.

(5) Specific volume of air at nozzle - cubic feet/lb.

(6) Air flow at nozzle - cfm.

(7) Barometric pressure - inches of mercury.

The equation for calculating air flow was derived as follows:

$V = 1{,}096 \quad hv$

$Q = 1{,}096\ C_d \cdot A \cdot \quad hv$

$1096 \cdot C_d \cdot A$ = constant for any given nozzle

For small 3.530-in. nozzle - $1{,}096 \cdot C_d \cdot A = 74.04$

For large 4.502-in. nozzle - $1{,}096 \cdot C_d \cdot A = 119.95$

Q for small nozzle = $74.04 \sqrt{hv}$

Q for large nozzle = $119.95 \sqrt{hv}$

Where: V = velocity at nozzle - ft/min
h = pressure drop at nozzle - in. of water
v = specific volume of air - cu ft/lb
Q = air flow - cfm
C_d = coefficient of discharge
A = area of nozzle

The specific volume, v, varies with wet bulb and dry bulb temperatures of air and the barometric pressure as determined from psychrometric charts.

c. Environmental Exposure. The frost test was performed in the calorimeter room. The air entering the evaporator section was maintained at 65° F for a period of 2 hours. The evaporator coil was then examined visually to determine if ice was present. Frost test on the 60-cycle unit was run in the USAERDL calorimeter room, and the 400-cycle unit test was witnessed at the contractor's plant.

The unit was tested for vibration in the vertical plane only at USAERDL. However, it was vibrated in a horizontal plane by a commercial laboratory under subcontract to the development contractor. Results are given in Appendix B. All preparation and operation of the USAERDL tests were done by personnel of the Packaging Development Branch. Tests were performed on an M-B Electronics Model C-25 vibration machine. The vibrating table was controlled from a console unit which also contained instrumentation to indicate amplitude and frequency of vibration. Control of the test was either automatic or manual depending on the complexity of the test and the limits of the automatic controls. Each section of the multi-package, 60-cycle unit was vibrated separately. The section under test was mounted on a fixture which was basically a large angle iron configuration with gussets welded across the ends. The back of the test specimen was bolted to the upright member of the fixture to simulate mounting for use. The weight of either section exceeded maximum limit of the vibrating machine; therefore, part of the weight was supported by an elastic rope. The elastic rope was supported overhead by a portable "A-frame" hoist.

The purchase description for the air conditioner called for a two-part vibration test. One part consisted of vibration of the unit at a constant amplitude of 3 G and at a frequency varying from 10 to 400 cps for 90 minutes. The other part consisted of vibration of the unit at the most pronounced resonant frequency with 3 G amplitude for an additional 90 minutes. The test at USAERDL was a three-part test; part one described above was divided into two parts, and the resonance test remained the same. The first part of the test conducted on each section was set up for automatic control and consisted of cycling up and down the frequency span of 70 to 400 cps for 60 minutes. The frequency range of 10 to 70 cps was scanned manually at a rate of 10 cps per 2-minute period with a 3 G input for the second part of the test. The duration of this portion of the test was 30 minutes. The unit was subjected to resonance test (third part) at 3 G input for 90 minutes or until a major failure was noted. The unit was then modified to enable it to withstand the required vibration. The subsequent tests to check the effect of different modifications consisted of scanning the frequency range of 10 to 200 cps to determine the severe resonant frequencies. When further vibration at these resonant points would obviously damage the unit, the test was discontinued and further modifications were made. If practical, the unit was vibrated at the three most pronounced resonant frequencies with a 3 G input for 30 minutes each. The testing and modifying were continued until favorable results were obtained.

The other environmental exposure tests and the tilted operation test were not performed or witnessed by USAERDL personnel. However, the 400-cycle unit which was retained by the contractor as a preproduction test item was subjected to all exposure tests which were witnessed by the Government inspector. The unit passed all tests.

7. Test Results. Test results were as follows:

a. Capacity. Tests at USAERDL indicated the capacity of the 60-cycle unit to be 9,480 Btu/hr at both 95° F and 125° F ambient temperatures. The power draw was 1,460 watts at 95° F and 1,750 watts at 125° F, and the respective coefficients of performance were 6.49 and 5.41 Btu/watt. USAERDL tests of the 400-cycle unit yielded a capacity result of 9,080 and 9,090 Btu/hr at 95° F and 125° F ambient temperatures. The power draw was 2,125 watts at 95° F and 2,250 watts at 125° F with the coefficients of performance being 4.28 and 4.04 Btu/watt, respectively. Summarized data for the USAERDL capacity tests is included in Appendices C and D.

Tests at ETL indicated the capacity of the 60-cycle unit to be 7,700 Btu/hr at 95° F ambient with a power draw of 1,490 watts and a resulting coefficient of performance of 5.17 Btu/watt. At 125° F ambient, the capacity was 8,800 Btu/hr with a power draw of 1,785 watts and a 4.93 Btu/watt coefficient of performance. The sensible heat factor was 0.795 at 95° F and 0.773 at 125° F. The ETL capacity tests of the 400-cycle unit gave a capacity of 8,200

Btu/hr at 95° F ambient with a power draw of 1,880 watts and a 4.37 Btu/watt coefficient of performance. At 125° F, the 400-cycle unit capacity was 7,200 Btu/hr with a power draw of 2,090 watts and coefficient of performance of 3.44 Btu/watt. The sensible heat factor was 0.738 and 0.784 at 95° F and 125° F, respectively. Data from ETL tests of both units is tabulated in Tables I and II.

Table I. Capacity Test Data, 60-Cycle Unit

Measurement	Low Ambient 95° F	High Ambient 125° F
Indoor (Evaporator Side) Calorimeter		
Air Temperature (°F)		
Room,[a] dry bulb	79.9	90.2
Room, wet bulb	67.0	75.2
Interspace, dry bulb (avg)	79.9	90.7
Humidifier Water Temperature (°F)	95.8	108.9
Weight of Water Evaporated (lb/hr)	1.4	1.9
Sum of Power Inputs to Calorimeter (watts)	2190	2460
Equivalent Heat Input (Btu/hr)	7480	8390
Heat leakage from Outdoor Calorimeter to Indoor Calorimeter through Separating Wall (Btu/hr)	140	315
Heat leakage from Interspace to Calorimeter through Walls, Floor and Ceiling (Btu/hr)	0	60
Outdoor (Condenser Side) Calorimeter		
Air Temperatre (°F)		
Room, dry bulb	95.2	125.0
Interspace, dry bulb (avg)	94.5	121.7
Cooling Water for Heat-Rejection Coils		
Coil No. 1		
Flow Rate (lb/hr)	1080	1015
Quantity Delivered (lb/hr)	1080	1015
Temperature (°F)		
Ingoing	53.5	61.2
Outgoing	62.8	75.1
Heat Removed (Btu/hr)	10040	14110
Coil No. 2		
Flow Rate (lb/hr)	880	845
Quantity Delivered (lb/hr)	880	845
Temperature (°F)		
Ingoing	53.7	61.2
Outgoing	63.0	73.5
Heat Removed (Btu/hr)	8180	10395
Heat Removed by Both Coils (Btu/hr)	18220	24505
Sum of Power Inputs to Calorimeter (watts)	1620	1785
Equivalent Heat Input (Btu/hr)	5525	6085

Table I (cont'd)

Measurement	Low Ambient 95° F	High Ambient 125° F
Heat Leakage from Outdoor Calorimeter to Indoor Calorimeter through Separating Wall (Btu/hr)	140	315
Heat Leakage from Calorimeter to Interspace through Walls, Floor and Ceiling (Btu/hr)	75	345
Air Conditioner		
Wet-Bulb Temperature of Air at Evaporator Grill(b) (°F)	57.7	68.0
Water Vapor Condensed (Equivalent to Water Evaporated from Indoor Calorimeter Humidifier) (lb/hr)	1.4	1.9
Electrical Input		
Test Voltage (Maintained) (volts)	115.0	115.0
Frequency (cps)	60	60
Line Current (amps)	14.3	16.9
E - Total Power Input (watts)	1490	1785
Energy Consumed (whr)	1490	1785
Power Factor (percent)	91	91
Miscellaneous		
Barometric Pressure (in. Hg)	29.78	29.86
Air Static-Pressure Difference between Indoor and Outdoor Calorimeters (in. H_2O)	0.000	0.000
Temperature of Refrigerant Circuit (°F)		
Entering Compressor	52.7	93.5
Leaving Compressor	131.0	240.0
Entering Expansion Valve	112.5	141.3
Leaving Evaporator	49.5	77.9
Refrigerant Charge (by Sight Glass)		
Condition	Clear	Clear
Moisture Content	Slightly Wet	Slightly Wet
Ratings(c)		
q_{t_r} - Net Total Room-Cooling Effect on Evaporator Side (Rated Capacity) (Btu/hr)	7700	8800
q_{t_a} - Net Total Room-Cooling Effect on Condenser Side (Btu/hr)	7800	8800
q_d - Net Room-Dehumidifying Effect (Btu/hr)	1500	2000
q_s - Net Total Sensible Cooling Effect (Btu/hr)	6200	6800

Table I (cont'd)

Measurement	Low Ambient 95° F	High Ambient 125° F
$\frac{q_{t_r} - q_{t_a}}{q_{t_r}}$ - Room-Cooling Effects Determined from Evaporator Side and Condenser Side (percent difference)	1.0	0
Btu/watt - Cooling capacity per watt of power input	5.17	4.93
Sensible Heat Factor	0.795	0.773

(a) At discharge of calorimeter air reconditioner.
(b) Measured - Wet-Bulb thermometer approximately 1/2 inch from grill.
(c) Ratings are reported to the nearest 100 Btu/hr since test methods, as specified in the "ASRE" Standard, do not warrant the reporting of ratings in increments less than 100 Btu/hr.

Note: Within the range of water temperature measured, the specific heat of water under constant pressure is taken to be unity. Therefore, differences in temperatures are substituted for differences in enthalpies in the computation of results.

Table II. Capacity Test Data, 400-Cycle Unit

Measurement	Low Ambient 95° F	High Ambient 125° F
Indoor (Evaporator Side) Calorimeter		
Air Temperature (°F)		
Room,[a] dry bulb	80.0	90.0
Room, wet bulb	67.0	75.0
Interspace, dry bulb (avg)	79.5	90.9
Humidifier Water Temperature (°F)	107.2	100.0
Weight of Water Evaporated (lb/hr)	1.9	1.3
Sum of Power Inputs to Calorimeter (watts)	2340	2005
Equivalent Heat Input (Btu/hr)	7990	6845
Heat Leakage from Outdoor Calorimeter to Indoor Calorimeter through Separating Watt (Btu/hr)	140	315
Heat Leakage from Interspace to Calorimeter through Walls, Floor and Ceiling (Btu/hr)	-60	110

Table II (cont'd)

Measurement	Low Ambient 95° F	High Ambient 125° F
Outdoor (Condenser Side) Calorimeter		
Air Temperature (°F)		
Room, dry bulb	95.3	125.1
Interspace, dry bulb (avg)	94.7	125.7
Cooling Water for Heat-Rejection Coils		
Coil No. 1		
Flow Rate (lb/hr)	1330	1335
Quantity Delivered (lb/hr)	1330	1335
Temperature (°F)		
Ingoing	57.0	66.2
Outgoing	64.2	76.6
Heat Removed (Btu/hr)	9310	15210
Coil No. 2		
Flow Rate (lb/hr)	1285	1290
Quantity Delivered (lb/hr)	1285	1290
Temperature (°F)		
Ingoing	57.0	66.2
Outgoing	65.4	75.8
Heat Removed (Btu/hr)	10410	12380
Heat Removed by Both Coils (Btu/hr)	19720	27590
Sum of Power Inputs to Calorimeter (watts)	1500	3885
Equivalent Heat Input (Btu/hr)	5115	13240
Heat Leakage from Outdoor Calorimeter to Indoor Calorimeter through Separating Wall (Btu/hr)	140	315
Heat Leakage from Calorimeter to Interspace through Walls, Floor and Ceiling (Btu/hr)	65	-65
Air Conditioner		
Wet-Bulb Temperature of Air at Evaporator Grill[b] (°F)	59.2	68.3
Water Vapor Condensed (Equivalent to Water Evaporated from Indoor Calorimeter Humidifier) (lb/hr)	1.9	1.3
Electrical Input		
Test Voltage (Maintained) (volts)	208	208
Frequency (cps)	400	400
Line Current (amps)		
Phase T_1	8.0	8.6
Phase T_2	7.8	8.4
Phase T_3	8.6	9.3

Table II (cont'd)

Measurement	Low Ambient 95° F	High Ambient 125° F
Power Input (watts)		
Phase T_1-T_2/T_1	210	390
Phase T_2-T_3/T_3	1670	1700
Total	1880	2090
Energy Consumed (whr)	1880	2090
Power Factor (percent)	60	68
Miscellaneous		
Barometric Pressure (in. Hg)	29.89	29.74
Air Static-Pressure Difference between Indoor and Outdoor Calorimeters (in. H_2O)	0.000	0.000
Temperature of Refrigerant Circuit (°F)		
Entering Compressor	71.0	84.5
Leaving Compressor	192.5	224.0
Entering Expansion Valve	105.0	134.0
Leaving Evaporator	59.0	68.5
Entering Evaporator	60.0	62.5
First Return Bend of Condenser	139.0	173.0
Mid Return Bend of Condenser	122.0	151.5
Last Return Bend of Condenser	113.5	144.5
Entering Receiver	109.0	139.5
Refrigerant Charge (by Sight Glass)		
Condition	Clear	Clear
Moisture Content	Dry	Dry
Refrigerant Pressure[c]	-	-
Ratings[d]		
q_{t_r} - Net Total Room-Cooling Effect on Evaporator Side (Rated Capacity) (Btu/hr)	8200	7200
q_{t_a} - Net Total Room-Cooling Effect on Condenser Side (Btu/hr)	8400	7400
q_d - Net Room-Dehumidifying Effect (Btu/hr)	2000	1400
q_s - Net Total Sensible Cooling Effect (Btu/hr)	6200	5800
$\frac{q_{t_r} - q_{t_a}}{q_{t_r}}$ - Room-Cooling Effects Determined from Evaporator Side and Condenser Side (percent difference)	2.5	2.5

Table II (cont'd)

Measurement	Low Ambient 95° F	High Ambient 125° F
Btu/watt - Cooling capacity per watt of power input	4.37	3.44
Sensible Heat Factor	0.738	0.784

(a) At discharge of calorimeter air reconditioner.
(b) Measured - Wet Bulb thermometer approximately 1/2 inch from grille.
(c) Gauges not provided with sample tested. Due to critical nature of refrigerant charge in a unit of this type, it was deemed inadvisable to attach gauges (unit factory charged); possible loss of charge by "bleeder-tap" gauges ruled out use of this method also. No values of pressure (arrived at by conversion of temperature to pressure) recorded; superheated gas temperatures preventing a direct comparison.
(d) Ratings are reported to the nearest 100 Btu/hr since test methods, as specified in the "ASRE" Standard, do not warrant the reporting of ratings in increments less than 100 Btu/hr.

Note: Within the range of water temperature measured, the specific heat of water under constant pressure is taken to be unity. Therefore, differences in temperatures are substituted for differences in enthalpies in the computation of results.

b. Air Flow. The quantity of air flow produced by the 60-cycle evaporator fan with dry evaporator coil ranged from 273 cfm to 96 cfm while the static pressure varied from 0 to 0.90 inch of water. At 0.15 inch static pressure, the air flow was 249 cfm. The same fan against a wet evaporator coil pushed air flow ranging from 258 cfm to 94 cfm as the static pressure was varied 0 to 0.9 inch. At 0.15 inch against the wet coil, the fan yielded a 242-cfm air flow. With the new TABLOCK fan wheel installed in the same unit, the air flow range increased to 312 cfm at 0 inch static pressure, 296 cfm at 0.15 inch, and 86 cfm at 1.65 inches. Curves for each of these tests are shown in Figs. 5 and 6.

c. Environmental Exposure. Visual inspection of the evaporator coil immediately after the frost test showed no severe ice formation and no impediment of air flow.

Examination of the evaporator section after parts I and II (cycling in the 10 to 400 cps frequency range at 3 G) of its vibration test showed that the mounting screws securing the fan

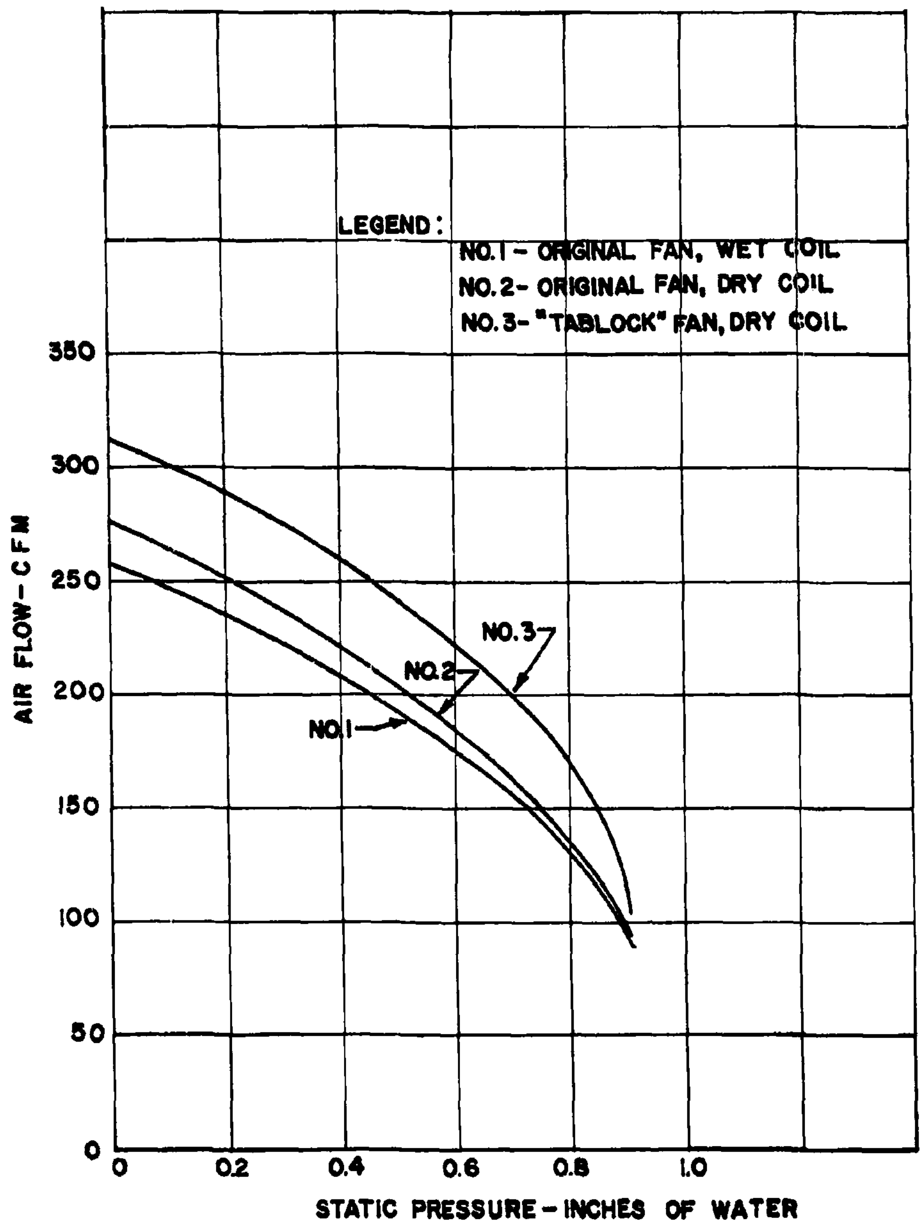

Fig. 5. Air flow vs static pressure, 60-cycle unit.

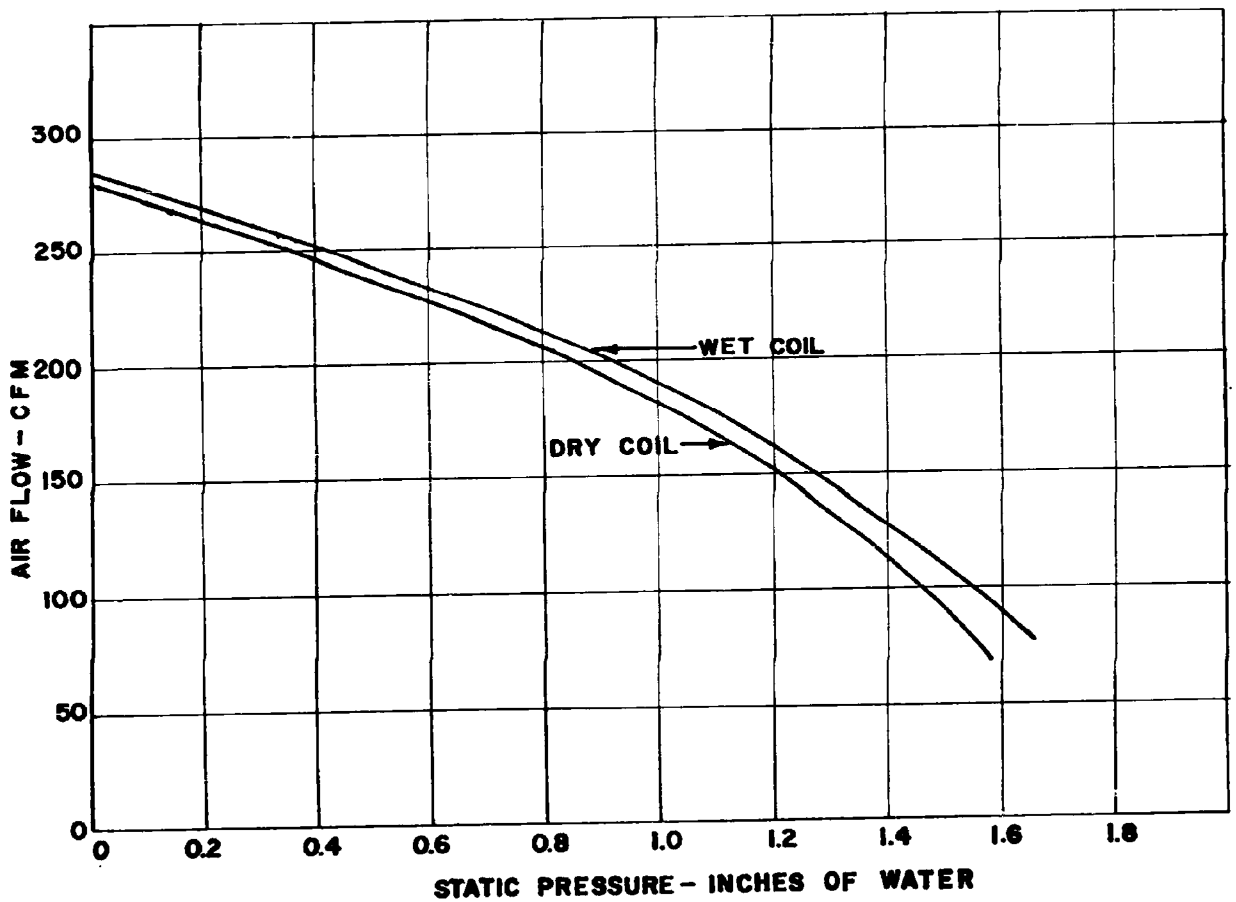

Fig. 6. Air flow vs static pressure, 400-cycle unit.

housing had backed out. Parts I and II of the test revealed that the most severe resonance of this section occurred between 31 and 32 cps. When the evaporator section was subjected to resonance tests at 31 to 32 cps (part III) for only 7 minutes, damage consisted of a broken weld joining the fan motor mounting base to the floor of the unit and an unhooked motor torque bracket. The condenser section was damaged after part I of its vibration test. The strap used to secure the fan motor to its base had broken, and a replacement strap broke during part II of the condenser section through the frequency range of 10 to 20 cps. Part III of the vibration test was not performed on the condenser section because severe damage would have been certain to occur.

The evaporator and the condenser sections were both subjected to modification and test by the trial and error method. The final tests were conducted with a wood block between the front of the test unit and the jig to eliminate the cantilever-type mounting.

Modification to the evaporator section included tightening of the motor torque mounts and bolting of the new motor mount base with radius bends and gusseted corners to the floor of the unit. When the unit was vibrated at resonant frequencies of 13.5, 17.5, and 24 cps for 30 minutes each, there was no apparent damage.

Modifications to the condenser section included addition of three reinforcing aluminum angles to the underside of the bottom pan assembly, a new condenser fan motor base with radius bend and gusseted corners, a new condenser fan motor strap of heavier gauge aluminum, and an angle joining the compressor discharge line and the compressor. The condenser section was vibrated at the resonant frequency of the compressor, 17 to 23 cps, with 2.5 to 3 G input for 19 minutes. The resonance was so severe that some resonance was induced to all other components and compressor displacement exceeded 1 inch. After 19 minutes, the compressor discharge line had broken and the external shock mounts were worn beyond effectiveness. The unit was then subjected to the resonant frequency of the fan, 28 cps, for 30 minutes, and no additional damage resulted. Examination of the inside of the hermetic shell of the compressor revealed that the internal discharge line was also broken and the internal shock mounts were badly worn.

III. DISCUSSION

8. Analysis of Test Methods. The capacity tests as performed at USAERDL failed to conform to ASRE Standard 16-56 because of inadequacies of the test facility. In the period following these tests, the calorimeter room was modified to produce accurate results. The amount of error introduced by deviations from the testing standard is

not known. The contractors who make and test the prototype air conditioners usually rate their unit higher than the results obtained in the USAERDL calorimeter. However, recent tests performed on the same test specimens by ETL yielded results even lower than the USAERDL results.

The facilities at ETL are constructed to the latest ASRE standard, and ETL does the testing of all commercial air conditioning units for American Refrigeration Institute's program of certified capacity rating. The ETL facilities were considered superior to the USAERDL facilities at the time the capacity tests were performed on the 9,000 Btu/hr multi-package air conditioner; for this reason, ETL's test results were included in this report.

The method used for measurement of air flow is also described in the standard. One source of inaccuracy in this test was the method of measurement of static pressure at the discharge of the air conditioner. The static pressure was measured in a straight transition (cross-sectional area equal to discharge opening of the test unit) rather than for the air to be discharged immediately into a plenum and the static pressure to be measured at that point. Measurement of static pressure at a plenum is more favorable since the almost negligible velocity pressure does not affect the measurement.

There are no inaccuracies in the frost test, since it involves operation only at 65° F with visual observation of the effects of the low temperature.

The vibration test also involves operation of the unit under certain conditions and visual observation of any effects. The purchase description for the unit describes the vibration requirements in an indefinite manner. Only the amplitude and frequency are spelled out, and no mention is made concerning the plane through which vibration should be applied or the method of mounting the air conditioner during the test. Recent discussions at USAERDL have indicated a desire to vibrate the air conditioners through one horizontal plane and the vertical plane. All tests performed at USAERDL included only vertical vibration of the unit, while the test performed by Inland Testing Laboratories included only horizontal vibration of the unit.

9. Analysis of Test Results. The 60- and 400-cycle units both failed to meet the purchase description capacity requirement of 9,000 Btu/hr. The 60-cycle unit produced only 7,700 and 8,800 Btu/hr at respective 95° F and 125° F ambients, whereas the 400-cycle unit produced only 8,200 and 7,200 at the respective ambients. The 60-cycle unit met the required 5.0 Btu/watt coefficient of performance with 5.17 Btu/watt at 95° F but failed to meet the requirement at 125° F when the coefficient of performance was 4.93 Btu/watt. The 400-cycle unit coefficients of performance were 4.37 and 3.44 Btu/watt at the

respective low and high ambients; both units failed to meet the required 4.5 Btu/watt. The sensible heat ratio was within the allowable range of 0.68 to 0.80 for all tests of each unit.

The 60-cycle and 400-cycle units yielded air flows of 273 and 285 cfm, respectively. These were the maximum air flows measured, and they did not meet the purchase description requirement of 300 to 350 cfm. Both units satisfied the condition of producing 85 percent of free air delivery at 0.15 inch of water back pressure with 249 cfm for the 60-cycle unit and 275 cfm for the 400-cycle unit. The air flow results of the 400-cycle unit revealed one serious inconsistency: that the wet coil air flow (285 cfm) was higher than the dry coil air flow (280 cfm). The wet coil result compares favorably with the 284 cfm obtained during the psychrometric calorimeter test and, therefore, is probably the more accurate. The new TABLOCK fan on the 60-cycle unit yielded 312 cfm at 0 inch back-pressure and 296 cfm at 0.15 cfm back-pressure. These results met both the free air delivery requirement and the 85 percent free air delivery requirement at 0.15 inch back-pressure.

The requirement that no ice form on the evaporator coil during low-temperature operation was met by both the 60- and 400-cycle units.

The unit subjected to vibration tests failed when the test was initially attempted. However, after a series of modifications, both sections of the unit withstood tests of resonant frequencies at 3 G input, except for the compressor and its discharge line. The discharge line was broken inside and outside the hermetic shell. The internal and external shock mounts were also badly worn. Studies on the shock and vibration problem are continuing to determine the specification requirements and means of meeting these requirements.

10. Evaluation of Units. Both the 60-cycle and the 400-cycle units failed to meet the purchase description requirements for capacity, air flow, and vibration; however, it is felt that these units perform as well as the standard single-package 9,000-Btu/hr air conditioner. Since coil frosting did not occur, air flow is considered adequate for both units. The design drawings have been modified to upgrade the design with respect to its ability to withstand vibration.

The face area of the multi-package unit is slightly larger, and the unit is 10 inches deeper than the single-package unit. The 60-cycle multi-package unit, therefore, could not replace the single-package unit where restricted mountings are involved. Bulkiness is the main disadvantage of the multi-package unit; however, the multi-package feature allows mounting with minimum modifications to the wall structure of the air conditioned space. Since this was the first development of a 400-cycle 9,000-Btu/hr air conditioner, it will not be required to replace an existing unit.

Although both the 60-cycle and the 400-cycle units meet the purchase description on sensible heat factors, they are considered too low for cooling electronic vans. Substitution of the new "Tab-lock" fan in the evaporator section would increase the sensible heat ratios to acceptable limits for cooling electronic vans. Experience has shown that the best flexible refrigerant hoses leak refrigerant to some extent; this makes recharging the unit necessary. This recharging is not desirable, and substitution of copper tubing for the hoses would eliminate the refrigerant loss.

Two of the basic purchase description requirements are no longer considered valid. These are the dual voltage feature (208/416 volts) and the storage compartment for the hose for connection of refrigerant lines for split mounting configurations. The original requirement for 416 volts appears to be based on early anticipation of Hawk Missile System requirements wherein the unit would have been mounted in a multiple stack on the ground support shelter end wall in a manner similar to some Signal Corps applications. Hawk now utilizes a single 38,000-Btu/hr unit. Change of the 9,000-Btu/hr multi-package air conditioner to a single voltage type (208 volt) will permit use of a single-voltage, standard-type compressor in both the 60- and 400-cycle versions, and electrical wiring harnesses and components will be simplified. The connecting refrigerant lines can be shipped inside the condensing unit without allocation of storage space for them. After a unit is installed, lines are never returned to the shipping space and the storage compartment is unused. The elimination of the dual-voltage feature and the hose storage space will permit redesign of a smaller and more economical unit.

IV. CONCLUSIONS

11. Conclusions. It is concluded that:

a. The 60-cycle and 400-cycle, 9,000-Btu/hr, multi-package air conditioning units do not meet all requirements set forth in the purchase description.

b. The 60-cycle and the 400-cycle units are suitable for applications where the multi-package feature is required.

c. There is no necessity for inclusion of the 416-volt operating requirement in design of the units.

d. The hose storage compartment on the unit is an unnecessary waste of space.

e. The multi-package design could be modified:

(1) To result in dimensions which more nearly correspond to those of the single-package unit.

(2) To eliminate the hose storage compartment.

(3) To eliminate the 416-volt components, controls, and wiring.

(4) To incorporate the "Tablock" fan in the evaporator section for increased air flow and sensible heat factor.

(5) To incorporate copper connecting lines in lieu of flexible hoses wherever installations will permit.

APPENDICES

APPENDIX A

AUTHORITY

Item No. 1420
CETC Mtg. #201

R & D/ SUB PROJECT CARD	TYPE OF REPORT: PROGRESS	REPORT CONTROL SYMBOL CSCRD-1(R1)
1. SUBPROJECT TITLE: HEATING AND AIR CONDITIONING EQUIPMENT, VAN TYPE (U)	2. SECURITY OF PROJECT: U; 4. INDEX NUMBER: 8-71-11-400	3. PROJECT NO.: 8-71-00-000; 5. REPORT DATE: 31 Dec 59
6. BASIC FIELD OR SUBJECT: Structures	7. SUB FIELD OR SUBJECT SUB GROUP: Heating and Air Conditioning for Prefab Structures	7A. TECH. OBJ.: SO-9
8. COGNIZANT AGENCY: C of E 9. DIRECTING AGENCY: Res & Dev Div, OCE 10. REQUESTING AGENCY: OCE	12. CONTRACTOR AND/OR LABORATORY: Engr Res & Dev Lab Harvey W. Hottell Ellis & Watts	CONTRACT/W. O. NO.: DA-44-009 eng-3378, 3312 DA-44-009 eng-3320, 3370
11. PARTICIPATION AND/OR COORDINATION: MED DEPT (C) ORD DEPT (C) QMC (C) CHEM CORPS (C) TRANS CORPS(C) CONARC (C) SIG CORPS (C) USN (C) USAF (C)	13. RELATED PROJECTS *: Nike-516-04-008(Hercules) Lacrosse-016-05-002 Corporal-016-05-005 Pershing 14. DATE APPROVED: 5 Aug 49 GSUSA 15. PRIORITY: 1-B 16. MAJOR CATEGORY: 5600	17. EST. COMPLETION DATES: RES. Cont.; DEV. Cont.; TEST Cont.; OP. EVAL. Cont. 18. FY. / FISCAL ESTIMATES: 60 — 379M; 61 — 240M; P/A — 400M
19. REPLACED PROJECT CARD AND PROJECT STATUS: Supersedes project card dated 31 Dec 58.		

20. REQUIREMENT AND/OR JUSTIFICATION

There is a requirement for an item capable of maintaining proper temperature and humidity levels in mobile or transportable enclosures in which equipment highly sensitive to extreme climatic conditions may be mounted. It is essential that map reproduction, communication, radar, and fire-control equipment be maintained at proper operating temperature if satisfactory performance is to be obtained. The project may result in items of materiel that possess such marked superiority over existing items that complete or extensive replacement would be justified.

21. BRIEF OF PROJECT AND OBJECTIVE

a. Brief:

(1) Objective:

Development of a combined heating-cooling-air conditioning unit, or series of such units, suitable for use with specialized van-enclosed equipment for map reproduction, communication, radar, fire control, and other similar purposes and capable of providing adequate operation conditions under all climatic conditions, and ambient temperatures ranging from 65° F. to 125° F.

* Hawk-516-04-006
Jupiter-516-05-010
Redstone-016-05-004

22. OASD (R & D)	SN.	CN.	C.	X.	I.	C.

DD FORM 1 APR 55 613 — PAGE 1 OF 3 PAGES
REPLACES DD FORM 613, 1 JAN 52.

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SUB R&D PROJECT CARD CONTINUATION SHEET

1420/201

1. SUBPROJECT TITLE	2. SECURITY OF PROJECT	3. PROJECT NO.
HEATING AND AIR CONDITIONING EQUIPMENT, VAN TYPE (U)	U	8-71-00-000
	4. INDEX NUMBER 8-71-11-400	5. REPORT DATE 31 Dec 59

Block 21a Continued

(2) Military Characteristics: See 31 Dec 57 Project Card.

b. Approach:

(1) A survey will be made of existing van-type enclosures and van-carried equipment, including equipment presently under development by various agencies. The result of the survey will be carefully evaluated in order that the problem of design and number of units required may be kept to a minimum.

(2) It is proposed to canvass industry and the military developmental agencies in order to determine if any product exists which can be used or modified to produce a satisfactory end item.

(3) If no item is available, research and development contracts will be awarded to outstanding industrial organizations with specialized engineering and scientific knowledge.

(4) Pilot models, based upon the most suitable design, will be procured and subjected to engineering tests, utilizing the facilities and personnel of ERDL and of the Army Industrial Hygiene Laboratory.

(5) Limited quantities, incorporating modifications indicated by engineering tests, will be procured for service test.

(6) Drawings and specifications suitable for quantity procurement will be prepared.

(7) Continuous liaison will be maintained throughout the development with all interested elements of the Departments of the Army, Air Force, and Navy.

c. Subtasks:

None.

d. Other Information:

(1) Scientific Research:

None.

(2) References:

(a) Memorandum from Logistics Division to OCE, subject: "Assignment of Research and Development Cognizance for Space Heating and Air Conditioning Equipment," dated 29 November 1948, file CSOLD/F1 412.11.

(b) Letter from Hq. AFF, Fort Monroe, Va., to Director of Logistics, GS, USA, subject, "Military Characteristics for Heating and Cooling Kits," file ATDEV-5 412.11/3 (17 Aug) dated 17 August 1948 with one inclosure and one indorsement.

(3) Discussion:

(a) The operation of equipment installed in mobile enclosures under extreme climatic conditions has created the need for special heating, cooling, and de-humidifying equipment if satisfactory performance is to be obtained. Tests at arctic sites and laboratory cold

SUB
R&D/PROJECT CARD
CONTINUATION SHEET

1420/201

1. [redacted] SUBPROJECT TITLE	2. SECURITY OF PROJECT	3. PROJECT NO.
HEATING AND AIR CONDITIONING EQUIPMENT, VAN TYPE (U)	U	8-71-00-000
	4. ~~INDEX NUMBER~~	5. REPORT DATE
	8-71-11-400	31 Dec 59

Block 21d Continued

rooms have indicated that existing conventional equipment is not capable of providing the degree of performance and reliability that is required.

(b) At present many components of map reproduction, communication, radar, and fire control equipment are inoperable under extreme climatic conditions and until new materials are available, methods of providing the required temperature and humidity control must be devised. This is essential if optimum performance is to be obtained.

(c) It is expected that the equipment developed under this project will be separated into the following functional components:

1. A basic unit to provide for air circulation and humidity control under all climatic conditions.
2. A cooling unit which may be used in conjunction with the basic component when cooling is required.
3. A heating unit which may be used in conjunction with the basic component when heating is required.

(d) Agencies interested in this project, in addition to the Corps of Engineers, are Department of the Navy, Department of the Air Force, CONARC, Signal Corps, Ordnance Department, Quartermaster Corps, and Medical Department.

DD FORM 1 FEB 56 613-1
REPLACES DD FORM 613-1, 1 FEB 53.

APPENDIX B

INLAND TESTING LABORATORIES
1482 Stanley Avenue
Dayton, Ohio

TEST REPORT

No. D-2193

for

ELLIS & WATTS PRODUCTS, INC.

on

Air Conditioner

Model CE-9B Sections 1 & 2

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ADMINISTRATIVE DATA

Ellis & Watts Products, Inc.
P. O. No. A 9697

1. NAME OF TEST ITEM: Air Conditioner (Two Sections, #1 & 2)

2. PURPOSE OF TEST: To perform a Shock Test and a Vibration Test in accordance with the detailed instructions from Ellis & Watts Products, Inc.

3. MANUFACTURER: Ellis & Watts Products, Inc.

4. MANUFACTURER'S MODEL NO.: CE-9B

5. APPLICABLE SPECIFICATION: None; Per detail instructions from manufacturer.

6. QUANTITY OF ITEMS TESTED: One unit composed of two sections.

7. DATE TEST COMPLETED: 22 August 1958

8. TEST CONDUCTED BY: Inland Testing Laboratories, 1482 Stanley Avenue, Dayton 4, Ohio.

9. DISPOSITION OF TEST ITEM: Upon completion of the tests, the sample was delivered to Ellis & Watts Products, Inc., 26 August 1958.

TABLE OF CONTENTS

SHOCK TEST

REQUIREMENTS:

One Air Conditioner, Model CE-9B (consisting of sections 1 & 2) shall be subjected to shock impacts of 15g intensity and 11 ±1 millisecond time duration, along each of 4 horizontal directions and the vertical down direction. Each section shall be tested separately.

TEST APPARATUS:

Shock Machine (Sand), per MIL-S-4456, ITL #818-D

TEST PROCEDURE:

The Air Conditioner (each section was shocked separately) was securely mounted to the shock machine table. Steel straps were placed around the case to assure rigid mounting. The table was then raised to a predetermined height and dropped, subjecting the sample to a shock impact of 15g intensity with a time duration of 11 ±1 milliseconds. Using the above procedure the sample was subjected to shock impacts along each of the 4 horizontal directions and once in the normal vertical down position. After each shock impact the sample was visually inspected for mechanical damage. For the diagram of the axes see Fig. 1 and 2 of Appendix I.

TEST RESULTS:

The visual examination disclosed no mechanical damage of the Air Conditioner as a result of the applied Shock Test.

TEST REPORT
1482 Stanley Avenue - Dayton, Ohio
Report No. D-2193

VIBRATION TEST

REQUIREMENTS:

One Air Conditioner, Model CE-9B (consisting of sections 1 & 2) shall be subjected to a vibration test at an amplitude of 3g, (1) scanning from 10 to 400 cps in 90 minutes and (2) vibration at the most pronounced resonance as determined in (1) above, for a period of 90 minutes.

TEST APPARATUS:

Vibration Machine, MB, Model C-5, ITL #817-D
Vibration Meter, MB, Model M3, ITL #806-D
Vibration Pickup, MB, Type 125, ITL #803-D
Flexure Table, ITL-#822-D

TEST PROCEDURE:

Each Air Conditioner section (each section was vibrated separately) was securely clamped to the flexure table, which was excited by the vibration machine. The sample was mounted in its normal position with the vibration applied through the "front to back" plane. The scanning for the most pronounced resonant frequency was conducted from 10 to 400 cps with a force of 3 g over a period of 90 minutes. Having determined the resonant frequency the sample was vibrated at this frequency for 90 minutes with a force of 3g. At the completion of the test the sections were visually examined for mechanical damage. The vibration axis is shown in Fig. 7 & 8 of Appendix I.

RESULTS OF TEST:

The resonant point of the evaporator section was at 196 cps. The resonant point of the condenser-compressor section was at 278 cps. There was no visible mechanical damage to either section as a result of the applied Vibration Test.

APPENDIX I

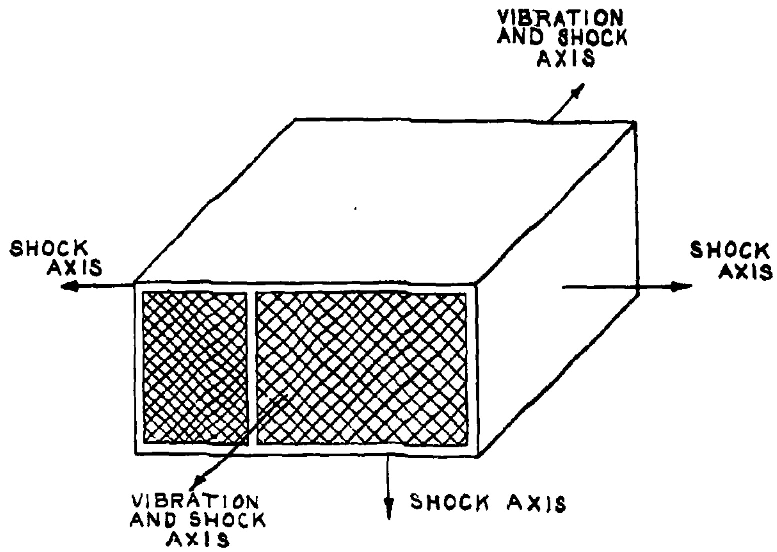

Fig. 7. Vibration and shock axes for compressor-condenser section.

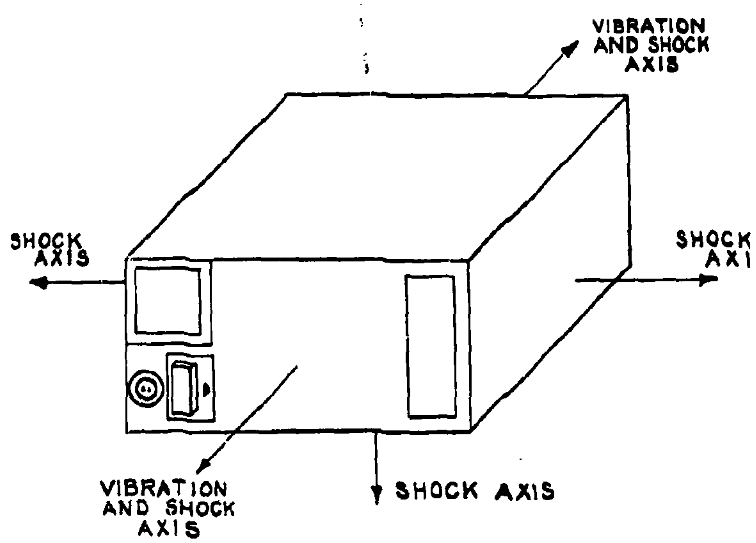

Fig. 8. Vibration and shock axes for evaporator section.

APPENDIX C

USAERDL CAPACITY TEST DATA, 60-CYCLE UNIT

Measurement	Low Ambient (95°F)[a]	High Ambient (125°F)[a]	High Ambient (125°F)[b]
Evaporator discharge air (°F)			
Dry bulb	63.2	70.3	73.6
Wet bulb	56.9	66.3	66.9
Room-side temperature (°F)			
Dry bulb	80.1	90.1	90.0
Wet bulb	67.4	75.0	75.3
Condenser discharge air (°F)	120.6	155.3	148.0
Outside (ambient) temperature (°F)	94.6	125.4	126.4
Laboratory temperature (°F)	73.6	78.0	75.0
Heat added to room side			
Sensible heat (watts)	1782	1555	1604
Latent heat (watts)	43	585	407
Fan motor (watts)	517	523	516
Lights (watts)	450	-	-
Total (watts)	2792	2663	2527
Total (Btu/hr)	9531	9091	8627
Heat loss or gain through walls (± Btu/hr)	-53	+392	+35
Net cooling capacity (Btu/hr)	9478	9483	8662
Power drawn by unit			
Volts	113	117	118
Amps	14.2	16.7	16.0
Watts	1460	1750	1700
Suction pressure (psig)	39	47	45
Coefficient of performance (Btu/watt)	6.49	5.41	5.09
Sensible Heat Factor	0.56	0.65	0.55

a. Evaporator section in room side; condenser section in ambient side.

b. Both sections in ambient side.

APPENDIX D

USAERDL CAPACITY TEST DATA, 400-CYCLE UNIT

Measurement	Low Ambient (95°F)	High Ambient (125°F)
Evaporator air discharge (°F)		
Dry bulb	61.5	71.7
Wet bulb	57.3	66.8
Room-side temperature (°F)		
Dry bulb	80.0	89.9
Wet bulb	66.7	74.8
Ambient-side temperature (°F)	94.3	124.5
Air temperature at nozzle (°F)		
Dry bulb	61.9	71.9
Wet bulb	57.7	66.9
Static pressure at test unit (inches of water)	0	0
Pressure drop at nozzle (inches of water)	1.921	1.232
Specific volume of nozzle air (cu ft/lb)	13.35	13.70
Air flow (cfm)	295	284
Enthalpy drop at test unit (Btu/lb)	6.84	7.1
Net capacity (Btu/hr)	9089	9083
Power drawn by unit (watts)	2125	2250
Supply voltage	213/213/213	211/211/211
Ampere draw	8.0/8.0/8.0	8.3/8.3/8.3
Coefficient of performance (Btu/watt)	4.28	4.04
Head pressure (psig)	160	236
Suction pressure (psig)	41	49
Sensible heat factor	0.70	0.68

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Report 1695-TR, 11 Sep 61, 43 pp, 8 illus, 2 tables
DA Project 8-71-11-400
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